

Effect of Valve Timing and Residual Gas Dilution on Flame Development Characteristics in a Spark Ignition Engine

Kang Xu, Hui Xie, Minggang Wan, and Tao Chen
Tianjin Univ.

Hua Zhao
Brunel Univ.

ABSTRACT

The goal of this research was to study and quantify the effect of exhaust valve timing and residual gas dilution on in-cylinder flow patterns, flame propagation and heat release characteristics in a spark ignition engine. Experiments were carried out in a recently developed single cylinder optical engine. Particle image velocimetry (PIV) was applied to measuring and evaluating the in-cylinder flow field. Detailed analysis of flame images combined with heat release data was presented for several engine operating conditions, giving insight into the combustion process in terms of visible flame area and flame expansion speed. Results from PIV measurement indicates that the limited alteration of the in-cylinder bulk flow could be observed with the variation of exhaust valve timing. The in-cylinder fluctuating kinetic energies and their Coefficient of Variations (COVs) decrease with the advance of the exhaust valve timing. Analysis based on the chemiluminescence imaging and the in-cylinder pressure shows that faster flame kernel formation and less distortion of flame shape in the early flame development stage could be obtained by the initial advance of exhaust valve timing, which benefits the combustion event by advancing the combustion timing and shortening the ignition delay and combustion duration without much increase in Coefficient of Variation of Indicated Mean Effective Pressure (COV_{IMEP}). By further advancing the exhaust valve timing to 32 CA Before Top Dead Center (BTDC), the combustion severely deteriorates with retarded combustion timing and higher cyclic variations, caused by the higher dilution level and lower fluctuation kinetic energy.

CITATION: Xu, K., Xie, H., Wan, M., Chen, T. et al., "Effect of Valve Timing and Residual Gas Dilution on Flame Development Characteristics in a Spark Ignition Engine," *SAE Int. J. Engines* 7(1):2014, doi:10.4271/2014-01-1205.

INTRODUCTION

Engines for passenger cars have to be operated under low load and low speed conditions during most of their lifetime. It is known that load adjustment in spark-ignition engines is traditionally realized by means of a throttle valve, which introduces additional pumping loss and significantly decreases the engine efficiency at part-load. Reduction of pumping loss can be realized by replacing part of the intake charge with burnt gas by means of optimizing the intake and exhaust valve timing combined with exhaust gas recirculation (EGR) and hence controlling the amount of internally recycled burnt gas and fresh charge [1, 2, 3, 4, 5]. Cycle Variations (CCVs), resulting deterioration of in engine efficiency and emissions [6, 7, 8].

Compared with external EGR (eEGR), internal EGR (iEGR) trapped in the cylinder is effective in charge dilution, because it provides not only volume dilution but also charge heating due

to its high temperature. The adoption of iEGR contributes in reducing not only the pumping loss [9], but also emissions of CO, HC and NOX, especially in direct injection engines [10]. However, high percentage of internal residual gas may lead to high ringing intensity and knock as well. There are several typical valve strategies to obtain internal residual gas, including exhaust gas re-breathing by positive valve overlap (PVO) [11, 12], residual gas trapping by negative valve overlap (NVO) [13, 14], and exhaust gas re-breathing by the secondary opening of exhaust during the intake stroke [15, 16].

In addition, iEGR plays an important role in the development of low temperature compression ignition combustion, such as HCCI and CAI. In order to extend the low temperature combustion operating range, spark ignition assisted SI-CAI hybrid combustion has been researched. In this case, combustion starts with flame propagation initiated by spark ignition followed by auto ignition combustion for the presence

of high iEGR. Experimental results by Persson etc. [17] show that flame propagation still prevails for high residual fractions achieved by NVO, and flame expansion speed decreases with higher NVO. But the relationship between fluctuations in combustion and flame propagation is still beyond explanation. Further research has to be carried out. In a SI engine operated in high dilution level, particularly diluted by both iEGR and eEGR, the flame propagation environment is much close to that in SI-CAI combustion. Therefore, the study of SI combustion operated in highly diluted conditions can be instructive in better understanding of the flame propagation in SI-CAI hybrid combustion.

The early flame development (corresponding to the Mass Fraction Burnt (MFB) 0~5%) is believed to deeply influence the subsequent combustion phase, therefore, any factors that accelerate this phase help to improve combustion stability [18, 19]. The main factors influencing the early flame development stage include turbulence intensity, local fuel/air ratio, spark ignition, residual friction, and thermal state, which are all directly or indirectly affected by valve actuations (valve timing, valve lift etc.)

The in-cylinder flow field is transient and varies from cycle to cycle. Test results by Enotiadis et al. [23] indicate that large scale velocity variations may not exhibit deterministic characteristics. Therefore, they are indistinguishable from turbulence. Cosadia et al. [24] studied the cyclic variations of swirling flow in a diesel engine. It is found that even the structure of the mean flow is complex and difficult to interpret because the phase averaged data contain only partly structural information. During the middle of the compression stroke, the swirling flow can be varying significantly from "vortex type" to "annular type" from one cycle to the other. The situation can be more complex in engines with variable valve actuations. Wang etc. studied the influence of maximum valve lifts (MVLs) on in-cylinder flow fields, including swirl flow, tumble flow and turbulence [25, 26]. The Particle Image Velocimetry (PIV) results indicated that with lower MVLs, stable tumble flow cannot be formed in the cylinder, whereas, higher swirl ratio and more stable swirl center can be achieved. The reduction in MVLs significantly increases the low-frequency fluctuating kinetic energy and leads to higher CCVs.

The exhaust valve actuations remarkably influence the amount of residual gas trapped in the cylinder. Fontana and Galloni studied the potential in fuel economy of variable valve timing in a small SI engine by 1-D and 3-D simulations [9]. By retarding the EVC timing, the exhaust gas was re-breathed into the cylinder during the intake stroke, and an intensive swirl motion of the re-aspirated exhaust at low valve lifts can be achieved owing to the special exhaust port geometry. Although

retarding EVC timing reduces the entire turbulence level, the turbulence near the spark is higher, improving combustion quality and toleration of charge dilution. The effects of trapped residual gas burnt (RGB) heterogeneity on early flame development in SI engines was studied by Cecile Pera etc.[27] with the application of Direct Numerical Simulation(DNS). Simulation results shows that RGB heterogeneity combines two opposing effects. Hot temperature pockets tend to locally increase the laminar flame speed, while diluted mixtures slow down flame propagation. And RGB heterogeneities are able to create additional flame wrinkling even without the introduction of sensible enthalpy fluctuations. However, the work of Pera et al. mainly focused on the effect of residual gas heterogeneity on the flame development at relatively small scale. In this paper, the influence of the global flame development considered, comprehensive study on the effect of exhaust valve timing on the in-cylinder flow pattern, flame propagation and combustion performance in SI engines was carried out.

The aim of this work is to investigate the impact of variable exhaust valve timing and iEGR dilution on the in-cylinder flow patterns, early flame development and combustion performance of SI engines by means of PIV measurements, and chemiluminescence imaging combined with in-cylinder pressure analysis. Firstly, the effects of EVC timing on in-cylinder bulk flow field and the cyclic variations were studied by PIV measurement in motored conditions, then a combined analysis of flame development and in-cylinder pressure traces in corresponding valve timings was carried out to bring the investigation of the early flame development and combustion stability into perspective.

EXPERIMENTAL SETUP

Engine Description

A single-cylinder engine with optical access to the combustion chamber was used for this investigation. Detailed information of the engine and the experimental setup are illustrated in [Figure 1](#) and [Table 1](#). This engine was modified from a production feasible four-cylinder gasoline engine. A single cylinder extension block was designed and installed between the original cylinder head and engine crank case. An elongated hollow piston was fixed onto the original piston. The optical access was provided by the piston window of fused silica glass with a diameter of 70mm as is indicated in [Figure 2](#). In addition, side windows in a metal block or a quartz glass ring could be used for additional optical access through the sides. Limited by its structural strength, the optical engine has to be operated at low speed and low load conditions. And the running time of the optical engine has to be strictly constrained in consideration of controlling the thermal load and the contamination of the quartz optical window.

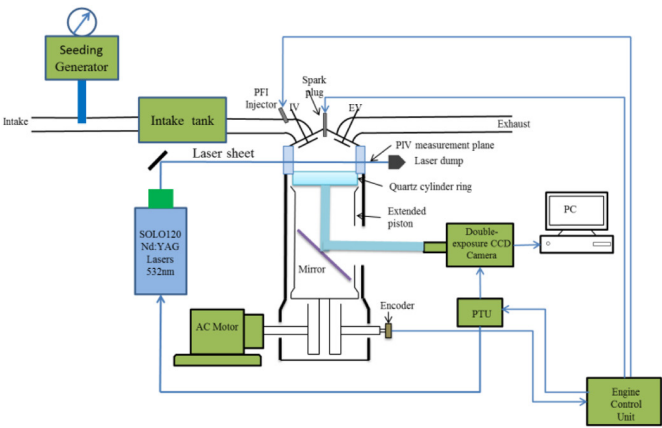


Figure 1. Schematic of the experimental setup

Table 1. Engine specifications

Engine type	4 stroke single cylinder
Bore	95mm
Stroke	95mm
Displacement	0.67L
Compression ratio	9.24
Combustion chamber	Pent roof/4 valves
Fuel injection	Port fuel injection
Fuel injection pressure	3 bar
Fuel	Gasoline 93 RON
Inlet pressure	Naturally aspirated
Coolant temperature	80±2°C
Engine speed	600r/min
Throttle	Non-throttled

Owing to the interchangeable camshafts, the valve timing and lift could be adjusted. For a given camshaft, the adjustment of valve timing was realized by changing the relative position of the crankshaft and camshafts through the timing chain.

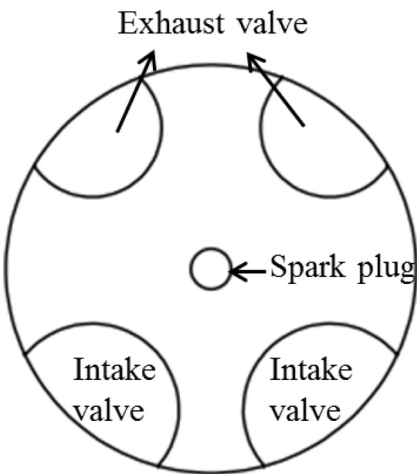


Figure 2. Visible area through the piston crown

PIV System and Flow Field Data Processing

PIV Measurement and Data Acquisition

The schematic of the PIV system is presented in Figure 1. FlowMaster PIV system developed by LaVision was employed for the measurements in this study. The second harmonic output (532 nm wavelength) from two SOLO120 Nd:YAG lasers was used as the light source for the PIV measurement. Firstly, the laser beam was sent through a spherical lens (1,000 mm focal length), followed by a concave cylindrical lens (20 mm focal length). Then, the laser sheet with a thickness of about 1.2 mm at the beam waist was formed. A Pulse Timing Unit (PTU) was used to control the laser timing and laser energy. When the PTU received an external trigger signal, it sent out four delayed signals according to the pre-set delay times, which were required to fire the flash-lamps and the Q-switches of the two lasers. Two laser pluses per PIV measurement were used to capture the in-cylinder flow image at assigned crank angle, with an interval of 20 μs. Laser sheet was introduced into the cylinder in the horizontal plane located about 10 mm below the electrode of the spark plug. For the PIV measurement, the engine speed was maintained at 600rpm and the in-cylinder flow field image was captured 50 degrees before the firing top dead center (BTDC_f), before the laser sheet was cut off by the moving upward piston. The reason why the certain crank angle is chosen is that our previous research has proved that the in-cylinder state at 50 degrees BTDC_f significantly influences the following combustion event [11].

A CCD camera (Sony ICX085 CCD sensor, Nikon AF Nikkor 50 mm f/1.8 lens) with an image resolution of 1,300 * 1,030 and 6.7μm pixel spacing was used for image acquisition. Seeding was added into the intake air flow by a Laskin sprayer with diisooctyl sebacate as solution. About 3-5 bar air in accordance with the requirement of seeding density was supplied to the sprayer by an air compressor. The average particle diameter ranges from 1μm to 3μm.

A DaVis 7.2 software package was used to acquire particle images and perform the post-processing. For processing of the PIV images, an interrogation spot size of 32 * 32 pixels was selected, which corresponded to a special resolution of 2.72*2.72mm on the objective plane. The average displacement for the particle pairs in each spot was calculated using a cross-correlation algorithm and was overlapped by 50 %. This spacing in the velocity field was small enough to describe flow features with a scale size as large as its integral length scale [28].

Calculation of Flow Parameters

The in-cylinder instantaneous velocity can be decomposed into the ensemble-averaged mean velocity and two fluctuating velocity components, including a low frequency large-scale component which is related to the cyclic variation of the large-scale flow, and a high-frequency small-scale component which is related to the turbulence [29, 30, 31, 32] as follows:

$$\vec{U}_{(x,y,\theta,i)} = \vec{U}_{EA(x,y,\theta)} + \vec{U}_{LF(x,y,\theta,i)} + \vec{U}_{HF(x,y,\theta,i)} \quad (1)$$

Where the ensemble-averaged mean velocity is calculated over the total cycles, N, by the following equation:

$$\vec{U}_{EA(x,y,\theta)} = \frac{1}{N} \sum_{i=1}^N \vec{U}_{(x,y,\theta,i)} \quad (2)$$

Where N is the total number of engine cycles. $\vec{U}_{(x,y,\theta,i)}$ is the two-dimensional instantaneous velocity vector at a point (x, y) at the crank angle θ of the *i*th cycle.

According to Li et al. [33], the fluctuating velocity is calculated by subtracting the mean velocity from the instantaneous velocity. Then the two components of the ensemble-averaged fluctuation intensity can be obtained from the root mean square (r.m.s) of the fluctuation velocity as follows.

$$u'_{EA(x,y,\theta,i)} = \sqrt{\frac{1}{N} \sum_{i=1}^N (U_{(x,y,\theta,i)} - U_{EA(x,y,\theta,i)})^2} \quad (3)$$

And

$$v'_{EA(x,y,\theta,i)} = \sqrt{\frac{1}{N} \sum_{i=1}^N (V_{(x,y,\theta,i)} - V_{EA(x,y,\theta,i)})^2} \quad (4)$$

Where (U, V) and (u, v) are respectively two components of the velocity vector and its fluctuation intensity in the x and y dimensions respectively.

The kinetic energy of the velocity fluctuation can be obtained from the two components of the fluctuation intensity as follows

$$E_{(x,y,\theta,i)} = \frac{1}{2} (u'^2_{EA(x,y,\theta,i)} + v'^2_{EA(x,y,\theta,i)}) \quad (5)$$

As the velocity fluctuation defined in the paper includes both low and high-frequency components, the fluctuation intensity and the kinetic energy can be derived in terms of low frequency, high frequency or total fluctuation values.

The decomposition of the in-cylinder instantaneous velocity is realized by a spatial low pass filtering scheme in the spatial frequency domain according to Li et al [28]. The two-dimensional instantaneous velocity $\vec{U}_{(x,y,\theta,i)}$ is firstly transformed into spatial frequency domain by a two-dimensional Fast Fourier Transform (FFT). Then an inverse FFT is performed after setting all Fourier coefficients above a given spatial cutoff frequency to zero. The cutoff frequency is determined by the Power Spectral Density (PSD) of the velocity fluctuation component $\vec{U}_{(x,y,\theta,i)}$ in each cycle. Detailed description of the PSD calculation can be found in the work of Liu et al [26].

Flame Image Acquisition

An intensified charge coupled device (ICCD) camera (Andor iStar DH734i-18F-03) with an UV-Nikkor 105 mm lens, which allows the ICCD camera to detect the luminescence in the UV and visible range from 180 to 880 nm, was used to record the chemiluminescence images of flame. A signal delay generator (Stanford Research DG645) controlled by the engine control unit was used to trigger the ICCD camera. The original images were digitized with 8-bit resolution with a size of 1024 × 1024 pixels. A graph paper attached to the piston crown was used to calibrate to optical resolution. It was found that the optical resolution of 1 pixel corresponded to a spatial dimension of 0.082 mm. Black paint was applied to the combustion chamber aiming at minimizing the impact of reflected chemiluminescence signal to obtain a negligible background signal.

Flame images were acquired in the following sequence. For each measurement point, 20 background images were firstly acquired prior to the spark ignition timing, which were averaged and then subtracted from the combustion image to improve the signal-to-noise ratio. After that, 20 chemiluminescence images were recorded at 13, 15, 17, 20 crank angle degrees (CAD) after spark ignition (ASI) respectively and then averaged to study the flame development characteristics. The number of flame images is limited by the running time of the optical engine and the framing rate of the ICCD camera (0.856 frame/second). It took about 24 seconds to finish each group of data collection. Simultaneously, 120 cycles of cylinder pressure were recorded to determine the CCV characteristics of combustion. Although 20 images for statistics are far from the usual CCV analyses, it is able to provide the information of flame development to some extent.

To obtain quantitative information from the flame images, a binarising-thresholding technique is applied for the determination of flame developing characteristics according to Aleiferis et al. [34].

In order to quantify the flame wrinkling, a parameter F_s (shape factor) is employed according to Gillespie [35]. It is defined as the ratio of the flame's perimeter to the perimeter of a circle with radius equal to the equivalent radius of the flame. The value of F_s is always no less than one, the higher the value of F_s , the more distortion of flame shape is presented.

A parameter A_{Norm} is defined as follows to quantify the relative value of the flame area:

$$A_{Norm} = A_f / A_{Visible} \quad (6)$$

Where A_f is the flame area, $A_{Visible}$ is 39.6 cm², the total visible area of the piston window.